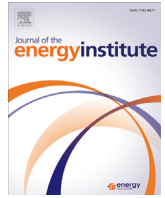




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Performance optimization of a linear phenomenological law system Stirling engine

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ABSTRACT

The power and efficiency performance analyses and optimization of a Stirling engine with heat resistance, heat leakage, regeneration loss and mechanical losses are carried out in this paper by using the combination of Senft's mechanical efficiency model with finite time thermodynamics analysis method. The analytical formulae for indicated power, shaft power, thermal efficiency and brake thermal efficiency for the Stirling engine are derived by assuming that the heat transfer at finite temperature difference between the heat reservoirs and the working fluid obeys the linear phenomenological heat transfer law. The optimal operating regions for shaft power and brake thermal efficiency are obtained and the influences of heat leakage, regeneration loss and mechanical friction losses on cycle performance are investigated through numerical calculations. The obtained results are expected to provide theoretical guidelines for the optimal design and operation of practical Stirling engines.

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1. Introduction

The Stirling engine is a simple type of external-combustion engine with good development prospects. It has the advantage of high efficiency, low vibration levels, simple structure and can run on any combustible fuels [1–3]. So, more and more attention has been paid to research work of Stirling engine. Kolin [4] had shown that with perfect regeneration, the thermal efficiency of an ideal Stirling engine can attain the maximum, i.e. the Carnot efficiency.

In recent years, the thermodynamic researches on the performance of the Stirling engine include many aspects. For example, one is the mechanical efficiency model developed by Senft [5–12]. Senft [5–12] studied the mechanical efficiency model of heat engine and applied to the performance analysis of the Stirling engine. He studied the characteristics of a fair-comparison class of reciprocating engines including Stirling engine and found that the ideal Stirling engine has the maximal mechanical efficiency. Thus taken together, the Stirling engine is seen to have the maximal brake-thermal efficiency in all reciprocating engines. Another is the finite time thermodynamic theory [13–22] developed by Andresen et al. Meanwhile, much work has been carried out on the finite time thermodynamic performance of the Stirling engine cycle and many new findings have been obtained [23–41]. Endoreversible Stirling engine model with heat resistance loss and irreversible Stirling engine model with losses of heat resistance, imperfect regeneration, heat leakage and internal irreversibility were built. The power, efficiency, exergoeconomic performance, ecological criteria and entropy generation of the common or solar-driven Stirling engines coupled to infinite or finite heat-capacity reservoirs with general working medium and quantum working medium were optimized.

Senft [39] examined the power and efficiency performance of the Stirling engine with heat resistance, internal heat leak and mechanical losses, in which heat transfer obeys the Newtonian heat transfer law $q \propto \Delta(T)$ by using the combination of his mechanical efficiency model [6–12] with the irreversible model in finite time thermodynamics [25,27,29,34]. The irreversibility of heat transfer is one of the most important facts in finite time thermodynamics. Many researchers [40–45] have shown that heat transfer law has a great influence on the optimal performance of endoreversible and irreversible Carnot and Stirling engine cycles. On the basis of Refs. [12,35,39], a further step

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List of symbols		λ	volume compression ratio
B	buffer space	η	efficiency
C	specific heat, W K	τ	cycle period, s
C_i	associated heat leakage coefficient, W K	<i>Subscripts</i>	
E	transmission force engine mechanism efficiency	B	buffer space
F	flywheel	c	compression
K_1	slope coefficient of temperature–time variation law in regenerator, K s ^{−1}	e	expansion
M	mechanism	F	flywheel
n	mole number, mol	H	hot side/high temperature heat reservoir
P	piston/power output of the engine cycle, W	L	cold side/low temperature heat reservoir
Q	the amount of heat transfer, J	m	optimal efficiency at maximum profit rate
R	regenerator	max	maximum value
R_g	universal ideal gas constant, J mol ^{−1} K ^{−1}	me	mechanical efficiency
T	temperature, K	opt	optimal value
t	time duration of the process, s	P	power/piston
V	volume, m ³	p	pressure
W	work, J	R	regenerator
x	working fluid temperature ratio	RL	regeneration heat loss
x_m	the optimal working fluid temperature ratio at maximum power output and maximum thermal efficiency	s	shaft
x_m^*	the optimal working fluid temperature ratio at maximum shaft power and maximum brake thermal efficiency	t	thermal
α, β	associate heat conductances, W K	0	ambient
		1	isothermal expansion process/hot side working fluid
		2	isothermal compression process/cold side working fluid
		3	regenerative heating process
		4	regenerative cooling process

made in this paper is to analyze the shaft power and brake thermal efficiency of the irreversible Stirling engine with heat resistance, heat leakage, regeneration loss and mechanical losses, in which the heat transfer between the heat reservoirs and the working fluid obeys the linear phenomenological heat transfer law $q \propto \Delta(T^{-1})$ by using the combination of Senft's mechanical efficiency model with finite time thermodynamics. The results obtained can provide theoretical guidelines for the design and optimization of practical Stirling engines.

2. Stirling engine model

The Stirling engine model working with two constant-temperature heat reservoirs is shown in Fig. 1. The temperatures of the high temperature and low temperature side heat reservoirs are T_H and T_L . The working fluid is assumed to be the steady-flowing ideal gas. Between the cold- and hot-side working fluids, there exists a regenerator R . Heat resistance, heat leakage and regeneration loss are considered here. However, some kinds of irreversibilities, such as intern heat rate losses transferred by fluid and solid conduction, pressure losses caused by internal friction and fluid escape are not considered in this model.

2.1. Linear phenomenological heat transfer law

Heat transfer between the working fluid and the heat reservoirs obeys the linear phenomenological heat transfer law. So, in case of isothermal expansion and compression Stirling engine process, the heat transferred into (Q_1) the engine from the hot heat reservoir and out of (Q_2) the engine to the cold heat reservoir per cycle are, respectively

$$Q_1 = \alpha(T_1^{-1} - T_H^{-1})t_1 = nR_gT_1 \ln \lambda \quad (1)$$

$$Q_2 = \beta(T_L^{-1} - T_2^{-1})t_2 = nR_gT_2 \ln \lambda \quad (2)$$

where α and β are the associated heat conductances between the engine and the reservoirs, T_1 and T_2 are the temperatures of the working fluid in the expansion and compression process, t_1 and t_2 are the time duration of the expansion and compression process, n and R_g are the mole number and the universal ideal gas constant of the working fluid, and $\lambda = V_2/V_1$ is the volume compression ratio. Here, Q_1 and Q_2 are supposed in absolute values.

3. Regeneration heat loss

Because of internal irreversibility, the efficiency of the regenerator is less than unity and the process is not an ideal regeneration process. So the regeneration heat loss is determined by [35,45]

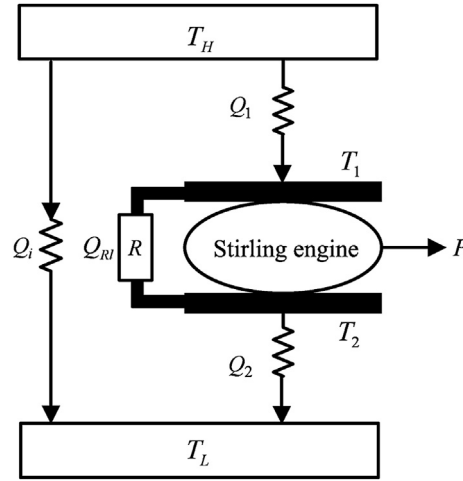


Fig. 1. Model of irreversible Stirling engine.

$$Q_{RL} = nC_v(1 - \eta_R)(T_1 - T_2) \quad (3)$$

where C_v is molar constant volume specific heat capacity of the working fluid and η_R is efficiency of the regenerator.

During the regeneration process, the temperature of the working fluid T has a linear variation law with the time duration [42]; if K_1 is the slope coefficient, it results:

$$dT/dt = \pm K_1 \quad (4)$$

where $K_1 > 0$ and is determined by the stuffing material of the regenerator. In Eq. (4), the symbol “ $\pm$ ” represents the regenerative heating (+) and cooling (–) processes, respectively.

4. Linear phenomenological heat leakage

The heat leakage between the heat reservoirs is [46]:

$$Q_i = C_i(T_L^{-1} - T_H^{-1})\tau \quad (5)$$

where C_i is the associated heat leakage coefficient and τ is the cycle period. It is due to the external conductance between the hot and cold heat reservoirs.

5. Thermodynamic optimal correlation between power output and thermal efficiency of Stirling engine

Combining Eqs. (1)–(5), the heats supplied by the hot heat reservoir (Q_H) and released to the cold heat reservoir (Q_L) are:

$$Q_H = Q_1 + Q_i + Q_{RL} \quad (6)$$

$$Q_L = Q_2 + Q_i + Q_{RL} \quad (7)$$

The cycle period is $\tau = t_1 + t_2 + t_3 + t_4$. Combining Eqs. (1) and (2) with Eq. (4) gives:

$$\tau = \frac{nR_g T_1 \ln \lambda}{\alpha(T_1^{-1} - T_H^{-1})} + \frac{nR_g T_2 \ln \lambda}{\beta(T_L^{-1} - T_2^{-1})} + \frac{2(T_1 - T_2)}{K_1} \quad (8)$$

Using the first law of thermodynamics and combining Eqs. (6)–(8) yields the power output (i.e. the indicated power) and the thermal efficiency of the Stirling engine:

$$P_i = (Q_H - Q_L)/\tau = (Q_1 - Q_2)/\tau = [nR_g(T_1 - T_2)\ln \lambda]/\tau \quad (9)$$

$$\eta_t = (Q_H - Q_L)/Q_H = (Q_1 - Q_2)/(Q_1 + Q_i + Q_{RL}) \quad (10)$$

Combining Eqs. (1)–(5) with Eqs. (8)–(10) gives:

$$P_i = nab\alpha R_g \ln \lambda K_1 / (2ab\alpha + ncR_g \ln \lambda K_1 T_2) \quad (11)$$

$$\eta_t = \frac{nab\alpha R_g \ln \lambda K_1 T_H T_L}{-2ab\alpha C_i \Delta T - ncR_g \ln \lambda K_1 T_2 C_i \Delta T + n\alpha b K_1 T_H T_L [R_g \ln \lambda + aC_v(1 - \eta_R)]} \quad (12)$$

where:

$$\Delta T = T_H - T_L, \quad a = 1 - x, \quad b = (T_2 - T_L)(xT_H - T_2), \quad c = T_2(T_H - \delta^2 x T_L) + T_H T_L(\delta^2 x^2 - 1) \quad (13)$$

and $\delta = (\alpha/\beta)^{0.5}$, $x = T_2/T_1$ is the working fluid temperature ratio.

In Eqs. (11) and (12), for fixed T_H , T_L , α and β , maximizing P and η_t respect to T_2 by setting $dP/dT_2 = 0$ and $d\eta_t/dT_2 = 0$ yields the same optimal temperature $T_{2,opt}$

$$T_{2,opt} = T_H T_L [T_H(1 + \delta x) - T_L \delta(1 + \delta)] / (T_H^2 - \delta T_L^2) \quad (14)$$

And the corresponding maximum power output P_i and maximum thermal efficiency η_t are, respectively:

$$P_i = \frac{-nmvR_g \ln \lambda K_1}{2(2vT_H T_L - r + \delta^{-1}l) - nmur \ln \lambda K_1 T_H T_L} \quad (15)$$

$$\eta_t = n\alpha h R_g \ln \lambda K_1 T_H T_L / \left\{ 2a\alpha C_i \Delta T (xd^2 + gT_H T_L + gdj) - ngR_g \ln \lambda K_1 T_H T_L C_i \Delta T (gl + dk) - n\alpha h K_1 T_H T_L [R_g \ln \lambda + aC_v(1 - \eta_R)] \right\} \quad (16)$$

where $d, g, h, j, k, l, m, r, u$ and v are expressed respectively, as:

$$\begin{aligned} d &= \delta^{-1} T_H^2 - \delta T_L^2, \quad g = T_H(\delta^{-1} + x) - T_L(x\delta + 1), \quad j = T_L + x\alpha^2 \delta^{-2} T_H, \quad h = (\delta T_L - T_H)(\delta^{-1} T_H - T_L)(xT_H - T_L)^2, \\ k &= 1 - \delta^2 x^2, \quad l = T_H - x\delta^2 T_L, \quad m = T_H^2 \delta^{-1} - 2T_H T_L + \delta T_L^2, \quad r = xT_H^3 \delta^{-1} - \delta T_L^3, \quad v = xT_H - T_L, \quad u = \delta x^2 + 2x + \delta^{-1} \end{aligned} \quad (17)$$

6. Analysis of mechanical loss [5–12]

A general reciprocating heat engine can be conceptually represented as Fig. 2 [5,12]. In the diagram, M represents the mechanism, F the flywheel and B the buffer space where the inside pressure of the working fluid action on the non-working space side of the power piston p_p . The source of buffer pressure is due to a special enclosure called buffer space. Usually, the buffer space is open to the surrounding atmosphere and thus the buffer pressure is the atmospheric pressure. The buffer gas absorbs stores and returns energy to the working fluid. It acts directly on the power piston and so it diverts and recycles some work W_{B+} away from the mechanism; the flywheel have a similar energy interaction W_{F+} . The flywheel and buffer pressure energy interaction reduces the mechanical friction losses of the engine.

The arrow in the diagram shows how the work transfer between the components. W_s is the useful work coming out of engine per cycle at the shaft engine. It is the difference between the cyclic work received by the buffer space (W_{B+}) and flywheel (W_{F+}) through the mechanism. And the cyclic work taken from flywheel (W_{F-}) and buffer space (W_{B-}) through the mechanism sustain the operation of the engine cycle. So, the amount of W_s depends on the transmission force engine mechanism efficiency (E). E is defined as the ratio of the actual output force to the ideal output force for any given input force applied [8,12].

The Stirling engine mechanical efficiency is the ratio of this cyclic shaft work W_s to output cyclic work W , which is the difference between the absolute expansion work W_e and the absolute compression work W_c of the engine cycle. For fixed volume compression ratio $\lambda = V_2/V_1$ and working fluid temperature ratio $x = T_2/T_1$, the cyclic mechanical efficiency can not exceed to [5–12]:

$$\eta_{me} = (E, x, \lambda) = E - (E^{-1} - E)S(x, \lambda) \quad (18)$$

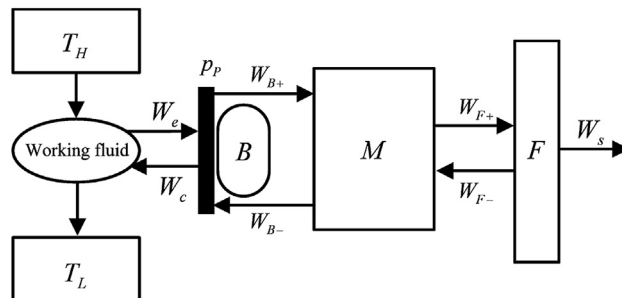


Fig. 2. Conceptual diagram of a reciprocating heat engine [5].

where:

$$S(x, \lambda) = \begin{cases} 0 & x\lambda \leq 1 \\ \frac{x \ln x - (1+x)[\ln(1+x) - \ln(1+\lambda)] - \ln \lambda}{(1-x)\ln \lambda} & x\lambda > 1 \end{cases} \quad (19)$$

The upper bound $\eta_{me}(E, x, \lambda)$ is the mechanical efficiency of an ideal Stirling engine buffered at its optimal constant pressure and having a mechanism of constant transmission force mechanism efficiency E .

7. Shaft power and brake thermal efficiency of Stirling engine

Combining Eq. (15) of Stirling cycle power output with Eq. (18) of the engine mechanical efficiency gives the shaft power of the engine:

$$P_s = P_i \eta_{me} \quad (20)$$

When the values of α , β , T_H , T_L , K_1 , λ and E are set, shaft power P_s is the function of working fluid temperature ratio x . Setting $dP_s/dx = 0$ gives the optimal temperature ratio x_m^* and the corresponding maximum shaft power. In the following, numerical examples are provided to analyze the optimal performance of the Stirling engine. In the calculations, ideal gas is chosen as the working fluid. It is set that $n = 1.0$ mol, $\alpha = \beta = 5.0 \times 10^6$ W K, $C_v = 20.77$ J/(mol K), $T_H = 800$ K, $T_L = 300$ K, $K_1 = 8.0 \times 10^3$ K/s and $\lambda = 4$.

Fig. 3 shows the shaft power P_s versus the working fluid temperature ratio x characteristic for different values of E . In the calculations, $C_i = 2.0 \times 10^4$ W K, $\eta_R = 0.8$ and $E = 0.8$ and 1.0 are set. The curves of P_s versus x are parabolic-like ones. From Eqs. (18) and (20), it can be inferred that when $E = 1.0$, the mechanical efficiency $\eta_{me} = 1.0$ and shaft power P_s equals the maximum indicated power P_i , as shown by curve 1 in Fig. 3. It follows from the figure that the maximum shaft power point can never exceed the maximum indicated power point, i.e. $P_{s,max} \leq P_{i,max}$ and $x_m^* \leq x_m$, with equality holding if and only if $1/\lambda \geq x_m$. So, one can see how mechanical losses reduce the output shaft power and decrease the value of optimal working fluid temperature ratio corresponding to the maximum shaft power.

Combining Eq. (16) of the Stirling cycle thermal efficiency with Eq. (18) of the engine mechanical efficiency gives the brake thermal efficiency of the engine:

$$\eta_s = \eta_t \eta_{me} \quad (21)$$

Fig. 4 shows brake thermal efficiency η_s versus the working fluid temperature ratio x characteristic for different values of E . In the calculations, it is set that $C_i = 2.0 \times 10^4$ W K, $\eta_R = 0.8$, $E = 0.8$ and 1.0 . The curves of η_s versus x are parabolic-like ones. When $E = 1.0$, the mechanical efficiency $\eta_{me} = 1.0$ and brake thermal efficiency η_s equals the thermal efficiency η_t , as shown by curve 1 in Fig. 4. It can be seen that the maximum brake thermal efficiency point can never exceed the maximum thermal efficiency point. As E decreases, the optimum temperature ratio x_m corresponding to the maximum thermal efficiency $\eta_{t,max}$ is slightly higher than x_m^* corresponding to the maximum brake thermal efficiency $\eta_{s,max}$. One can see that the temperature ratio x_m corresponding to the maximum brake thermal efficiency in Fig. 4 is different from the temperature ratio x_m corresponding to the maximum shaft power in Fig. 3. In Eqs. (11) and (12), T_2 and x are set as two variables. Their exist the same optimal temperature $T_{2,opt}$ which lead to the maximum output power P_i and the maximum thermal efficiency η_t . Both P_i and η_t are functions of temperature ratio x . For a heat engine, the maximum power operation regime and the maximum efficiency operation regime are usually attained at two different working points, which lead to the different values of x_m in Figs. 3 and 4.

Fig. 5 shows the influence of heat leakage on shaft power P_s versus the brake thermal efficiency η_s characteristic. In the calculations, it is set that $\eta_R = 0.8$, $E = 0.8$ and $C_i = 0, 2.0 \times 10^4$ W K and 5.0×10^4 W K. When there is no heat leakage in the cycle ($C_i = 0$), the curve of P_s – η_s is a parabolic-like one, as shown by curve 1 in Fig. 5. There exists the optimal brake thermal efficiency which leads to the maximum shaft power corresponding to the maximum power operation regime. When there exists heat leakage in the cycle ($C_i = 2.0 \times 10^4$ W K and $C_i = 5.0 \times 10^4$ W K), the curves of P_s – η_s are loop-shaped ones, as shown by curves 2 and 3 in Fig. 5. Both the brake thermal efficiency and the shaft power of the engine have a maxima corresponding to the maximum power operation regime and the maximum efficiency operation regime [47]. The maximum shaft power $P_{s,max}$ will not be affected by the heat leakage. $\eta_{s,max}$ and $\eta_{s,opt}$ decrease while $P_{s,opt}$ increases as heat leakage increases.

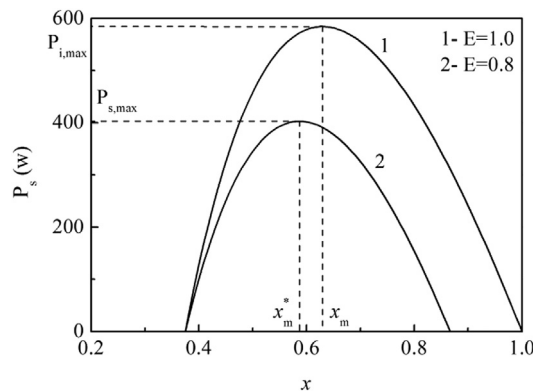


Fig. 3. Shaft power P_s versus working fluid temperature ratio x for different E .

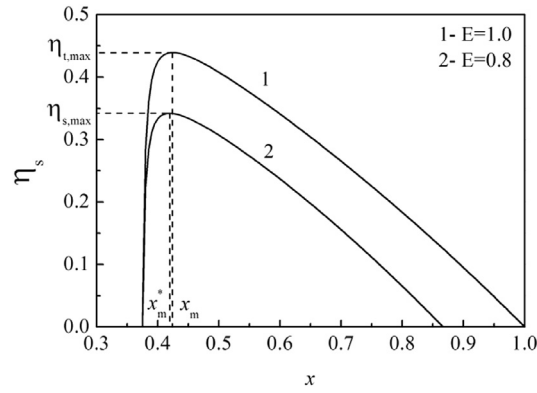


Fig. 4. Brake thermal efficiency η_s versus working fluid temperature ratio x for different E .

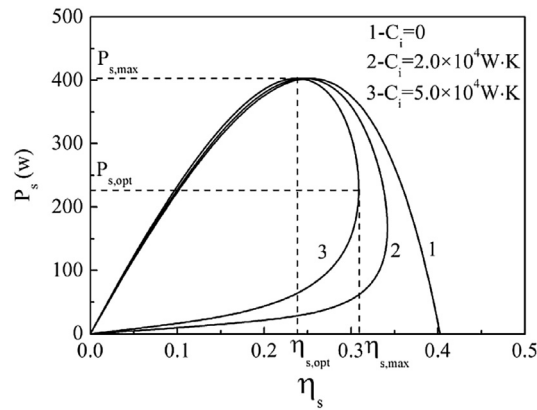


Fig. 5. Influence of heat leakage on the shaft power P_s versus the brake thermal efficiency η_s characteristic.

The optimal operation regions for the irreversible Stirling engine with heat leakage should be located in the regions where the P_s – η_s curve has a negative slope. Thus, the optimal shaft power and brake thermal efficiency of the engine should be:

$$P_{s,opt} < P_s < P_{s,max} \quad (22)$$

and

$$\eta_{s,opt} < \eta_s < \eta_{s,max} \quad (23)$$

where $P_{s,max}$, $P_{s,opt}$, $\eta_{s,max}$ and $\eta_{s,opt}$ are important performance parameters which determine the lower and upper bounds of the optimal shaft power and brake thermal efficiency.

Fig. 6 shows the influence of the regeneration heat loss on the shaft power P_s versus the brake thermal efficiency η_s characteristic. In the calculations, $C_i = 2.0 \times 10^4$ W K, $E = 0.8$ and the efficiency of the regenerator is set as $\eta_R = 0.9$, 0.8 and 0.7 , respectively. The corresponding

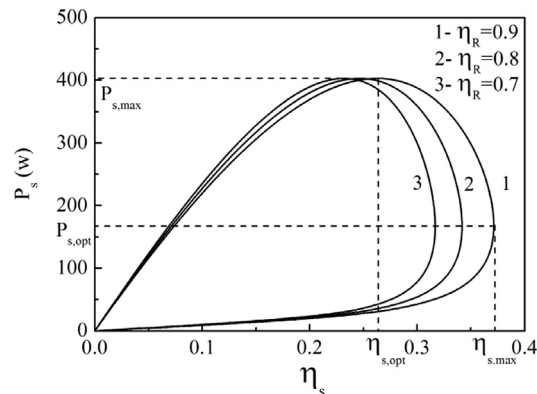


Fig. 6. Influence of regeneration heat loss on the shaft power P_s versus the brake thermal efficiency η_s characteristic.

curves are shown by curves 1, 2 and 3 in Fig. 6. As regeneration loss increases, the maximum efficiency $\eta_{s,max}$ and the optimal efficiency $\eta_{s,opt}$ decrease; while the maximum and optimal shaft power $P_{s,max}$ and $P_{s,opt}$ will stay unchanged.

8. Conclusion

The power and efficiency performance analyses and optimization of a Stirling engine are carried out in this paper by using the combination of Senft's mechanical efficiency model with finite time thermodynamics analysis method. The model built in this paper represents several typical kinds of losses of an ideal Stirling engine: heat transfer loss, heat leakage, regeneration loss and mechanical loss. Specially, the heat transfer of the engine model between the working fluid and the heat reservoirs obeys the linear phenomenological heat transfer law $q \propto \Delta(T^{-1})$. The analytical expressions of shaft power and brake thermal efficiency are derived. Numerical examples show that heat leakage, regeneration heat loss and mechanical losses have great influences on the power and efficiency performance of the Stirling engine. The optimal operation regions for power and efficiency have been determined. The results can provide some guidelines for the design and optimization of the Stirling engine.

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